Acceleration Effects on Electron Tubes

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This paper discusses methods of measuring shock and vibratory accelerations to which electron tubes may be subjected in various equipments, and the influence of these disturbances on the performance of the tubes. An outline is given of the design problems connected with the elimination of tube damage or faulty operation of tubes under adverse shock or vibration conditions, and of the methods used for simulating these conditions by means of production test machines and test methods.

INTRODUCTION

The rapid expansion of the use of electronic equipment by industry and the armed services has created increasingly new demands on both the electrical and mechanical characteristics of electron tubes. Since these tubes are electronic devices, it is only natural that their structural designs are dictated to a large extent by electrical requirements. However, the experience gained with conventional tubes in some of the new equipment applications has revealed certain mechanical shortcomings which reflect on the proper functioning or life of the tubes. The realization of the increasing importance of mechanical design has resulted in an increased effort for structural improvements to assure more reliable tube performance. The term "reliable", as used here, denotes that a tube has a high degree of dependability when subjected to specific conditions, either electrical or mechanical. Thus, the requisites for reliability may differ for various applications. Although, in designing a tube, many requirements must be taken into consideration, only some of the problems connected with dependable tube performance under mechanical disturbances, i.e., shocks and vibrations, will be discussed here.

Since the performance of a tube depends on the geometry of its component parts, minute changes in element spacing may produce variations in its characteristics. Because of the necessarily delicate structure of some tube elements, permanent or transient dimensional changes may be produced by mechanical forces acting on the tube unless the tube is

specifically designed to minimize tbe effects of such distrubanees. For a rational design, it is, therefore, necessary to have some knowledge of the nature of the disturbances likely to be encountered by tubes during their service life. If one considers the numerous conditions under whicb electronic equipment is required to function, it becomes clear tbat the mechanical requirements are manifold indeed. Equipment applications can be divided into three general groups, each one imposing in many cases special requirements on tubes. These groups are: (a) stationary equipment, such as central office telephone installations, home radio and television receivers, etc., (b) mobile equipment used in land vehicles, ships or airplanes, and (c) portable equipment. In many instances, military equipments straddle above subdivisions and superimpose additional requirements. It must be stressed, too, that pre-service conditions encountered through handling and shipping must be taken into consideration, since a tube is of no use to the customer until it is installed and operating.

A knowledge of expected service conditions is not only useful in the initial design stage but also aids the manufacturer in devising suitable test gear to check the quality of the product at the factory. Although a wealth of data has been collected on shocks and vibrations to which electronic equipments are subjected under actual service eonditions, little is known how these disturbances are altered by the mechanical structures of the equipments before they reacb the tubes. In general, the nature and magnitude of mechanical disturbances can be expressed in terms of acceleration, velocity, or displacement. Since electron tubes may respond to a wide frequency range of vibrations, the most sensitive measure is acceleration, which varies as the square of the frequency of the element displacement. Velocity or displacement instruments are usually not sufficiently sensitive to give a true record of disturbances in the higher modes of vibration due to the small velocity and displacement values involved.

The investigation of tbe nature of accelerations at tube sockets offers special problems. The accelerometers must approximate tbe weights of the tubes used in the respective sockets so that the disturbances are not modified by the substitution of acceleration pick-ups for the tubes. For the same reason, the method of fastening the accelerometers in tbe sockets must duplicate that of the tubes, and since the accelerating forces may act in several directions, the accelerometers must be capable of exploring these various directions. Lightweight accelerometers meeting the above requirements have been developed recently. These instruments are generally built to approximate tbe weight, weight distribution, and shape of vacuum tubes. A more detailed description of these accelerometers and associated recording circuits is given in the following chapter.

ACCELEROMETERS AND ASSOCIATED INSTRUMENTATION

Instrumentation employed for measuring accelerations depends largely on the frequency range of the disturbances to be recorded, which, for vacuum tube applications, covers approximately the audio spectrum.

In practically all cases, electronic recording equipment is used because of its high degree of flexibility. Its basic components consist of (a) a mechanical-electrical transducer which translates mechanical disturbances into proportional electrical potentials, (b) an electronic amplifier to step up the voltage output of the transducer to desired levels, and (c) an indicating instrument, usually a cathode ray oscilloscope, for observations of the disturbances.

Accelerometers

Although several types of mechanical-electrical transducers or pickups are in existence,[1] it has been found that the self generating types employing materials such as quartz or ferro electric crystals, are most useful since they can be constructed so that their output is directly proportional to acceleration over a large frequency range. Velocity and displacement indicating instruments are sometimes used for low frequency work; by suitable differentation their output signal will be proportional to acceleration.

An accelerometer must have the following fundamental properties:

(a) the signal produced must be proportional to the accelerations to be measured.

(b) its calibration must be stable and unaffected by humidity and temperature changes encountered.

(c) its mechanical strength has to be adequate to withstand the acceleration stresses to which it is subjected.

(d) its weight must be sufficiently low so that the disturbance patterns are not modified by loading.

(e) its sensitivity (voltage output/g) and useful frequency range must be sufficiently high to cover acceleration magnitudes and frequency components to be recorded.

A number of piezoelectric materials are available and have been employed in various electro-mechanical transducers. The development of practical light-weight transducers has been made possible by the use of some of the relatively new ferro-electric ceramics. By proper compounding, firing and poling, these materials can be made to have very high sensitivity and good life stability. Other advantages are their high dielectric values, good mechanical strength and relative insensitivity to usually

encountered humidity and temperature conditions. Accelerometers approximating the size and weight of electron tubes are now commercially available and used for measuring accelerations imparted to tubes through their sockets.

By proper construction the voltage output of these pick-ups is proportional to the stresses induced in the active elements within the desired frequency range. Depending on choice of material and construction, their active elements can be used in compression, shear, or torsion. Fig. 1(a) shows the simplified construction of a compression type pick-up. It consists of (1) a base which is rigidly fastened to the point at which accelerations are to be measured, (2) a sensitive element, (3) a weight, and (4) a retaining spring. The whole assemhly is shown held together by a screw. The mechanical equivalent of this structure is shown in Fig. 1(b).

When this unit is suhjected to sinusoidal motion its displacement is given by

$$X = X_0 \cdot \sin \omega \cdot t$$

Where X = instantaneous displacement of the base from equilibrium at time t

X_0 = maximum displacement from equilihrium

ω = circular frequency of the motion

The instantaneous acceleration then becomes

$$a = \frac{d^2x}{dt^2} = -X_0\omega^2 \text{ Sin } \omega t \tag{1}$$

The motion is transmitted to the mass M through the parallel spring

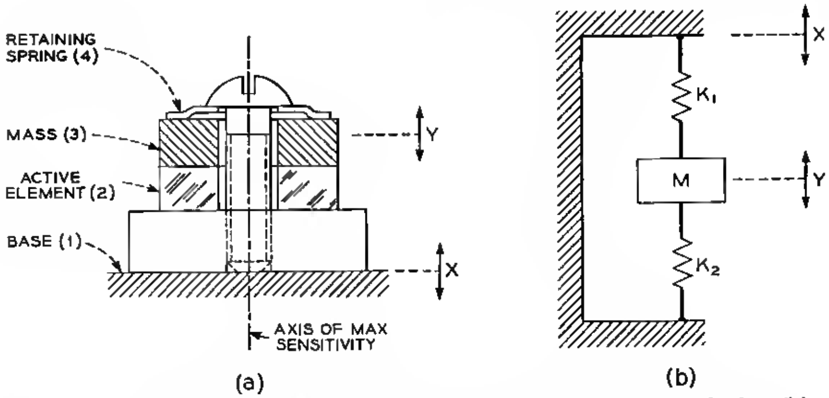


Fig. 1 — Compression type accelerometer (a) and its mechanical equivalent (b).

system formed by the stiff active element K_2 and the retaining spring K_1. For negligible damping in these springs the motion of the mass relative to the base is:

$$y_1 = \frac{X_0 \left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_n}\right)^2} \cdot \mathrm{Sin}\ \omega t \tag{2}$$

Here ω_n = the natural circular frequency of the mass on the springs. The force F exerted by the springs on the mass is:

$$F = (K_1 + K_2)y_1 = \frac{(K_1 + K_2)X_0 \left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_n}\right)^2} \mathrm{Sin}\ \omega t \tag{3}$$

The accelerometer is constructed so that the spring constant K_1 of the retaining spring is considerably smaller than that of the active element K_2. Therefore, for vibration frequencies relatively low as compared to ω_n, equation (3) becomes:

$$F = \frac{K_2}{\omega_n^2} X_0\omega^2 \mathrm{Sin}\ \omega t$$

Where (4)

$$\frac{K_2}{\omega_n^2} = M, \qquad \text{and} \qquad X_0\omega^2 \mathrm{Sin}\ \omega t = a$$

The stress produced by this force F produces a charge on the active element which is proportional to the instantaneous value of the acceleration to which the unit is subjected. Properly calibrated therefore, the acceleration can be measured in gravitational units. When the disturbing force consists of a number of these components, the total instantaneous output is proportional to the sum of these components, within the frequency limitation of the pick-up.

In deriving the above expressions, a number of assumptions have been made. Equation (4) therefore holds only if:

(a) ω_n is made large compared to the highest shock or vibration frequencies that are to be recorded.

(b) K_1 is small compared to K_2. Since the function of the retaining spring is merely to hold the assembly together for peak accelerations encountered, a soft spring with a sufficiently large static deflection can be employed or the spring may be replaced by conducting cement to hold the parts together.

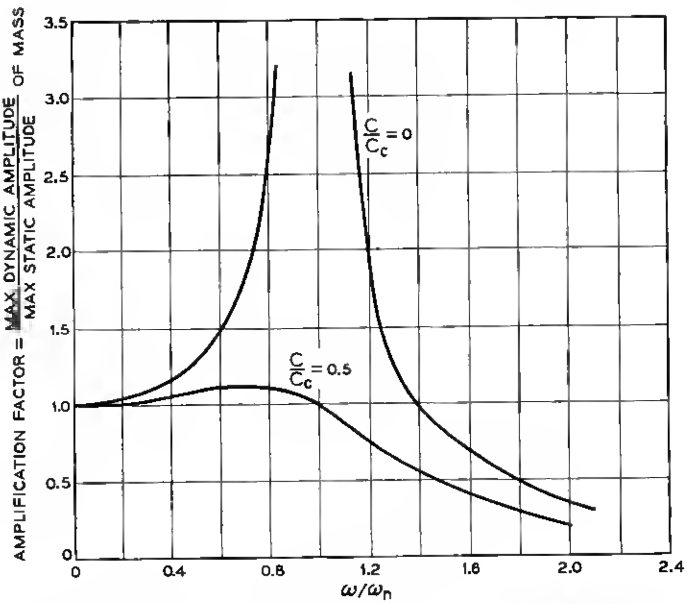


Fig. 2 — Amplification of forced vibration on mass for two degrees of damping.

(c) damping of the system is small. If damping is considerable, equation (2) becomes

$$y_1 = \frac{\left(\frac{\omega}{\omega_n}\right)^2 X_0 \operatorname{Sin} \omega t}{\left(\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2 \frac{c}{c_c} \frac{\omega}{\omega_n}\right]^2\right)^{1/2}} \quad . \tag{5}$$

In this expression c/c_c = fraction of critical damping. However, since by construction, the natural frequency of the system is large compared to the forcing frequencies, and damping of the active elements employed in these accelerometers is small, the term $\left(2 \frac{c}{c_c} \cdot \frac{\omega}{\omega_n}\right)^2$ becomes negligible. Equation (5) therefore simplifies to equation (2).

The useful upper frequency limit is generally fixed at $\frac{1}{4}\, \omega_n$, because, as shown by Fig. 2* amplification of the impressed signal results as ω/ω_n increases. If the disturbances contain frequencies beyond the useful limit, low pass electrical filters must be employed in the associated amplifier to suppress these frequencies. The fact that this type of pick-up has very low damping offers some disadvantage in recording complex waves be-

* Figs. 2 and 3 were obtained from material given in Reference 2.

cause phase distortion between the impressed and recorded frequency components takes place. Fig. 3* graphically illustrates this relation. This figure shows that $\left(\frac{c}{c_c}\right)$ would have to be made approximately 0.5 in order to have a phase angle proportional to the impressed frequencies, i.e., to obtain a true recorded pattern of the disturbance. Damping is generally not employed in accelerometers built to duplicate the size and weight of electron tubes due to constructional difficulties. The useful frequency limit of these devices is often not governed by the resonant frequencies of the active elements but by the lowest resonant frequency of their housing structure.

As mentioned previously, the active elements of these light weight transducers can be employed in various ways. The first models which were developed a few years ago made use of the elements in compression, following the practice of their predecessors, or: the relatively heavy and large quartz crystal accelerometers. A working model of a compression unit is shown in Fig. 4(a). Although this type of unit can be constructed to have a very wide useful frequency range (high resonant frequencies), it possesses two disadvantages: its internal capacity is rather low, and it has poor directional sensitivity. Its sensitivity decreases to roughly

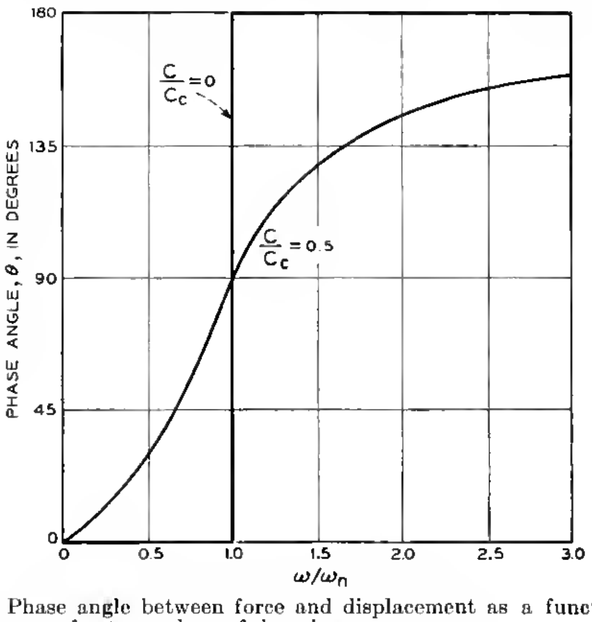


Fig. 3 — Phase angle between force and displacement as a function of the forcing frequency for two values of damping.

10 per cent of maximum in the direction perpendicular to the sensitive axis. Some improvements in directivity were obtained by utilizing the elements in shear. Fig. 4(b) illustrates a tandem shear type accelerometer. The construction of this unit is somewhat more complicated. The elements first have to be poled in the proper direction. The conducting layers used for poling in this direction then have to be removed and new conductive coatings have to be applied on the areas facing the base and mass M. The unit shown is sensitive to accelerations in the radial direction. The improved directivity of the shear elements over compression elements is shown in Fig. 5. Although shear elements are superior for detecting acceleration components along desired directions, it is difficult to make their internal capacity sufficiently high without increasing the over-all weight of the accelerometer for a given sensitivity. The use of cantilevers for these applications also suggested itself, for relatively high tensile and compressive stresses can be produced in such structures. Cantilever type elements have been constructed as shown schematically in Fig. 6(a). Here thin laminations of active material are cemented on opposite sides of a small metal cantilever. Under the action of external forces bending of the inner member subjects the outer laminations to

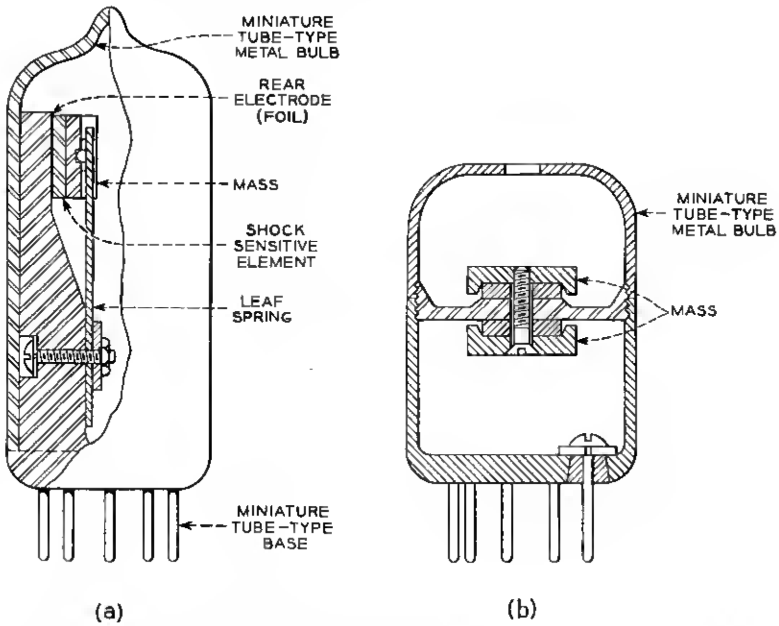


Fig. 4 — (a) Radial miniature tube compression type accelerometer and (b) radial miniature tube shear type accelerometer.

tension and compression stresses respectively, thereby producing the desired charges in these elements. The characteristics, i.e., the resonant frequency and sensitivity, of these elements are largely determined by the dimensions of the metal member. Relatively high internal capacities can be obtained by this construction. A recent variation of such a structure is shown in Fig. 6(b). In this design the inner metal component of the cantilever described above has been eliminated. The entire structure is made of the active material, in this case in cylindrical shape for ease of

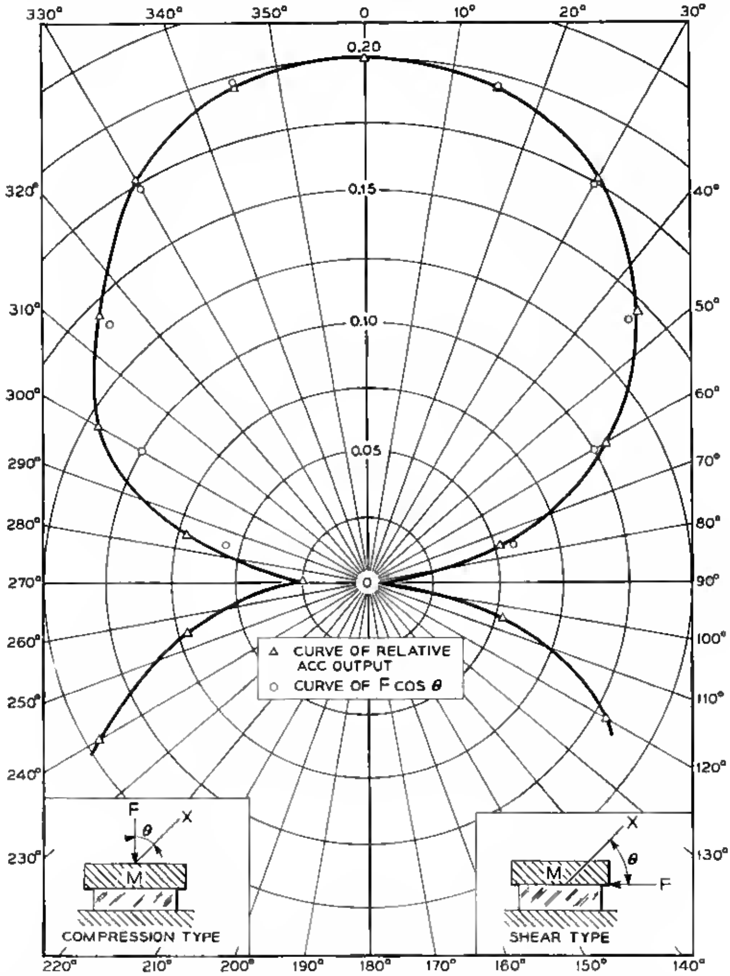


Fig. 5 — Relative angular response of compression and shear type ferro electric crystal accelerometers.

manufacture. Sections of conductive coating are deposited on this cylinder as shown. These are used to initially polarize the material, and, in use, to collect the charges produced by stressing the inner and outer fibers of the cylinder. It is interesting to note that these elements can be used to detect, by proper choice of connections, either radial or axial accelerations. Through suitable external instrumentation both directions can also be recorded simultaneously if desired.

The above illustrations show but a few applications of this material for detecting accelerating forces. Additional forms will suggest themselves, each of particular advantage for specific application.

The sensitivity of an accelerometer is generally given in coulombs/g or open circuit voltage. Its effective voltage output over a given frequency range is a function of the characteristics of the associated equipment including the connecting cable. Calibration in the lower frequency range up to a few hundred cycles may be performed by vibrating the accelerometer on variable frequency vibration machines or resonating spring

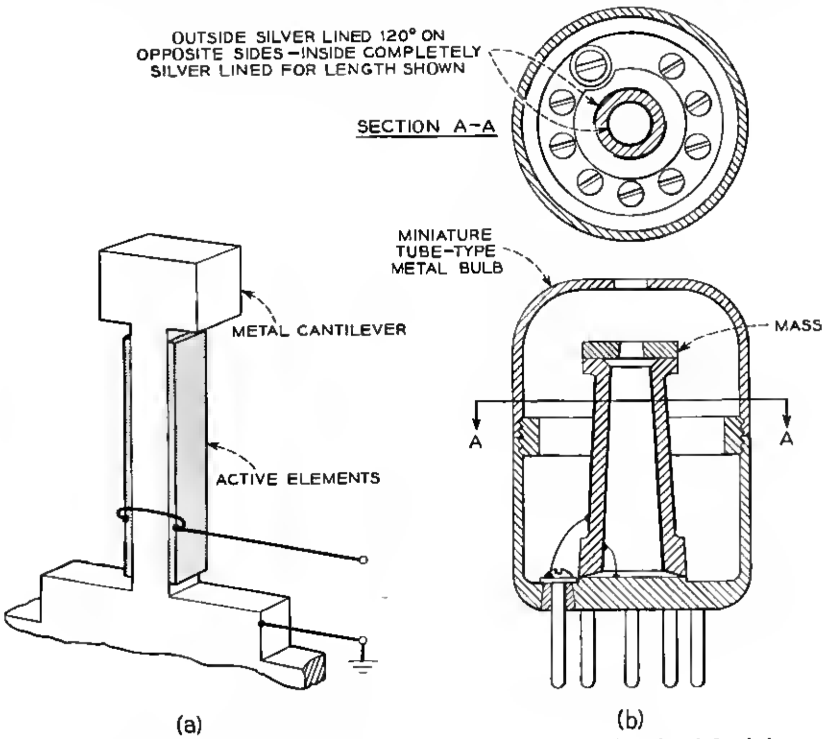


Fig. 6 — (a) Metal cantilever type element and (b) radial and axial miniature cantilever type accelerometer.

systems at amplitudes that are measurable with the help of a microscope. The peak acceleration in gravitational units for sinusoidal motion is given by:

$$g = .102\ (\text{c.p.s.})^2 \times \text{amplitude (inches)}$$

At higher frequencies, this method of calibration becomes increasingly difficult due to the decrease in obtainable amplitudes. A calibration method for a wide frequency range is described in Reference 3. In general, the sensitivity of an accelerometer is a function of the active material, its size, and the weight M employed. The useful frequency range increases with decreasing sensitivity. For a given design, a compromise therefore has to be made between these quantities and the over-all permissible weight of the finished unit. To illustrate the approximate relations of weight, size and sensitivity, units have been constructed in the shape of miniature tubes weighing only 33 per cent more than their prototypes, with sensitivities in the order of 0.005V rms/g and a useful frequency range of 3,000 cycles.

Since these accelerometers are calibrated for rectilinear motion, the results obtained in measuring equipment vibrations or testing machines must be carefully interpreted. In many instances the disturbing forces impart a rocking motion to the units so that the position of the active elements in the housings will influence the acceleration magnitudes that are registered.

Associated Instrumentation

Since the voltage output of self generating accelerometers is small, electronic amplifiers have to be employed to bring the signal to desired levels. Fig. 7 illustrates a typical arrangement of the necessary equipment. A cathode ray oscilloscope is shown as the visual indicating means, although other recording instruments can be employed, depending on the nature of the disturbances to be measured and the type of record desired. The prime requirements of the equipment are:

(a) phase distortion must be low, so that the disturbance pattern is correctly presented.

(b) its transient and frequency response must be adequate for the frequencies to be recorded.

In the circuit shown in Fig. 7, the signal is fed into a cathode follower stage having a high input impedance in order to obtain good sensitivity and frequency response. The use of a cathode follower also offers more flexibility in the proper matching of the high impedance pick-ups to suitable low pass filters or to the following amplifying stage. The generated

signal is shunted by the capacity of the connecting cable. Since the voltage impressed on the grid of the first stage is approximately

$$V_2 = V_1 \frac{C_1}{C_1 + C_2}.$$

it is desirable to have C_1 large compared to C_2. In some types of investigations cable whip may result in the introduction of spurious voltages on the signal. These voltages are produced by capacitance changes and static charges on the cable dielectric. To minimize this effect, the cable may be shunted by a padding condenser at the expense of the voltage V_2. The National Bureau of Standards recently reported the development of a low noise cable[4] that is reported to overcome the shortcomings of the ordinary shielded cables. Cables with a sufficiently low noise figure have also become available commercially.

A timing and calibration voltage can be impressed across the low valued resistor R for comparison with the disturbance signal. If resonant frequency signals of the accelerometers are excited by the disturbances being investigated, or the recorded frequency range is to be limited, the signal is fed through low pass filters before being amplified. This prevents possible overloading of the amplifier by the unwanted frequencies, so that maximum gain can be realized for the desired signal.

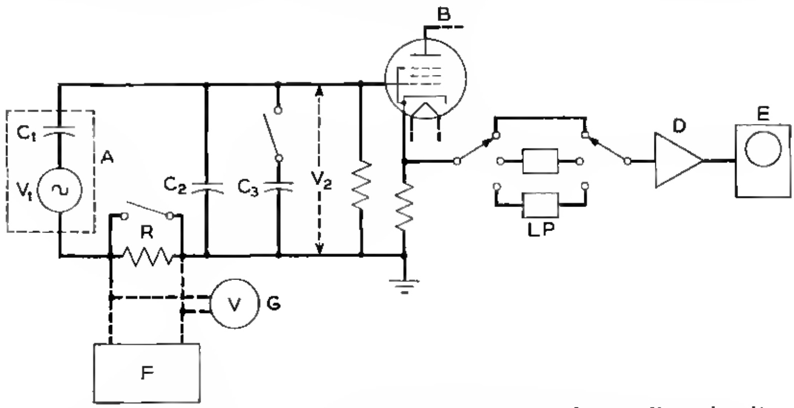


Fig. 7 — Schematic of accelerometer and associated recording circuit.

A — Equivalent circuit of accelerometer
B — Cathode follower stage
LP — Low pass filters
C_1 — Internal capacity of accelerometer
C_2 — Cable capacity
C_3 — Padding condenser
D — Voltage amplifier
E — Recording instrument
F — Audio oscillator
G — Voltmeter
R — Calibrating resistor
V_1 — Voltage source
V_2 — Voltage impressed on first stage

Particular attention must be paid to the design of the filters since these elements are apt to introduce excessive phase shift and oscillations. A variable frequency electronic filter has been found to be quite serviceable due to its flexibility in the choice of cut-off frequencies.

Additional information on shock and vibration instrumentation is given in Reference 5. Since the presentation of mechanical distrubances is, to some extent, limited by the transducers and their associated circuits, there is a trend towards a certain amount of standardization of these components so that results can be compared on an industry wide basis.

TUBE DESIGN PROBLEMS IMPOSED BY ENVIRONMENTAL CONDITIONS

It follows from the numerous acceleration measurements made at the installation points of equipment, that electron tubes, together with other components, have to withstand a large variety of conditions. Without over-simplifying the problems involved, it is perhaps permissible to divide these disturbances into two classes: ballistic shock, and transitory or sustained low g vibrations. Although, as a first approach, the effect of these distrubances on tube elements may be probed by mathematical analysis, the final design must be proven in by laboratory tests under controlled conditions. In discussing the influence of shock and vibration on tubes, their elements are frequently presented schematically as cantilevers, Fig. 8(a). While this assumption is a close approximation for some of the older tubes, such as the Western Electric No. 349B tube, Fig. 8(b), most tubes of later design, Fig. 8(c), do not lend themselves to such simple analysis due to their more complex structure. The response of elements to even simple shock pulses on tube envelopes are influenced by factors such as the clearances in micas, mica fits in the bulbs and tightness of mount assemblies.

It is obviously beyond the scope of this paper to analyze the destructive effects of shocks and vibrations on equipment. A few of the many excellent articles and publications on this subject are listed in Reference 6. It is equally impossible to present all of the many problems facing the electron tube engineer in designing tubes that will reliably serve their purpose under adverse conditions. Since tubes must be designed to withstand distrubances encountered in the field or be adequately protected, the following notes, highlighting some of these prohlems, will be of interest to equipment as well as tube engineers.

Influence of High g Shocks

In certain applications, tubes are required to withstand occasional high shocks such as those produced in military applications by explosions

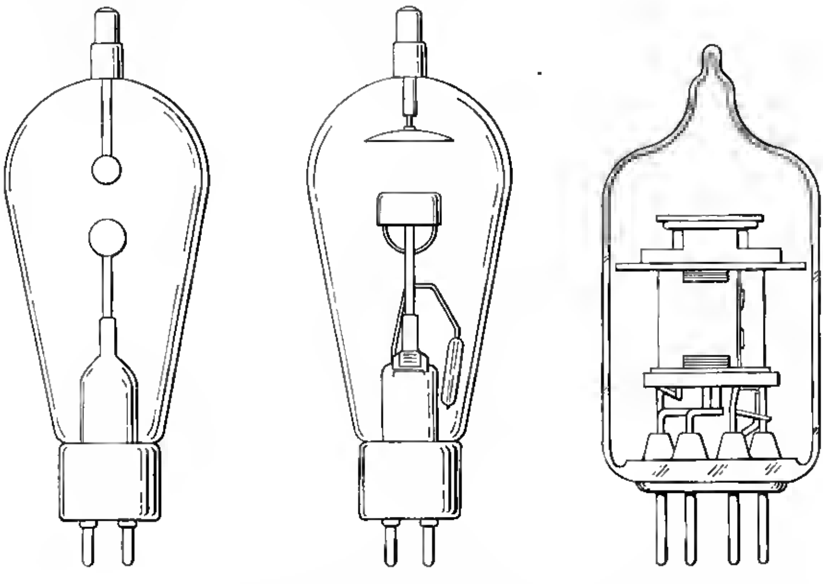
of some kind. It is generally permissible that, during very severe shocks, electronic equipment is non-operative; therefore, the response of tubes during these disturbances may fall outside of assigned limits. However, since it is highly important that operations resume immediately, the tubes must show no permanent change nor have caused damage to the circuits as a result of temporary faulty operation. Although protective shock mounts are usually employed on equipments exposed to these conditions, damped equipment vibrations excited by the attenuated shock wave are superimposed on the pulse felt by the tube.

Brittle or stiff tube components are most susceptible to shock because of their inability to absorb the shock energy by elastic deformations. Failures falling into this category are:

(a) glass breakage, which may be brought about by impact due to excessive movement of the tube or adjacent components during the impact.

(b) metal to glass seal fractures, produced by shock loads on the tube leads and seals.

(c) heater or filament failures and opening of welds.



(a) (b) (c)

Fig. 8 — Electron tube structures: (a) Simplified anode and cathode structure of a Western Electric No. 349B tube, (b) Western Electric No. 349B tube, and (c) typical miniature tube.

(d) damage to mica serrations and enlarging of the mica holes which support and space tube elements. The deterioration of the mica is also known to liberate gas, which, in turn, results in a reduction of the vacuum.

Permanent damage is also caused by deformation of elements beyond their elastic limit. Bowing of grids and cathodes as a result of shock has been reported in some applications.

Although not always realized by design engineers, shocks and vibrations of surprising magnitudes may also occur in handling and shipping. If these conditions are not taken into consideration during the design stage, the over-all cost of the product may be adversely affected by the necessary protection that has to be built into the shipping container to assure safe arrival of the packaged article at its destination. While many factors enter into the proper design of shipping containers, such as moisture and corrosion protection, the selection of cushioning materials is perhaps the most important. Since it is desirable from a storage and shipping cost standpoint to keep the package bulk to a minimum, and protective packaging cannot always compensate for design weakness, adequate strength must be designed into the tube even though it will not be subjected to severe shocks once it is installed. A rather complete analysis of the dynamics of package cushioning is given in Reference 7. At present, military requirements specify that packaged tubes must safely withstand several three foot drops onto a hard surface.

Influence of Low g Disturbances

In contrast to the relatively infrequently occurring high peak shock and vibrations, we find that many equipments are often subjected to repetitive shocks and sustained vibrations at lower acceleration levels. These conditions are generally encountered in vehicle, ship and airplane applications. Although these shocks and steady state vibrations can be attenuated through the use of shock and vibration mounts, the effectiveness of these mounts may be reduced sharply by a change in disturbance frequencies from normal.

Tube failures resulting from these conditions are generally caused by fatigue of some tube elements. For instance, the continual hammering of micas against tube walls or chattering of cathodes and grids in the mica, may reduce the value of the micas as supporting and spacing elements, and since tubes are required to function under these conditions, the gradual degradation of the micas will bring about an increase in the tubes' microphonic output. Where microphonism is a factor, the useful

life of a tube is, therefore, terminated long before more apparent failures occur in the tube structure. Other points of weakness are heater and filament leads.

As may be deduced from the above, microphonism is a frequent cause of complaint when tubes are subjected to low but repetitive shocks or transient vibrations. To illustrate the influence of such disturbances on tube performances, the results of recent investigations of tube microphonism found in a certain equipment will be cited.

In this case, field reports indicated that certain tubes exhibited excessive microphonism in the equipment, although the tubes were found to be within limits when judged by standard factory tests. It was apparent, therefore, that these tests did not simulate actual conditions. Acceleration measurements made at the equipment base and at tube sockets revealed that the steep, short duration impact of a blow delivered to the outer case excited resonant vibrations of the chassis on which the tubes were mounted. The magnitudes of these vibrations were only in the order of 0.1g, but their lowest frequency (approximately 550 cps) was close to the mount resonances of some of the tubes. Further tests also showed that those tubes having pronounced response, i.e., low damping, to vibratory motion in this frequency range also proved to be microphonic in the equipment. (The vibration spectrum of one of the tubes is reproduced in Fig. 9.) It was found that the various modes of vibration of the tube mount produced the high peaks in the range between 500 and 1000 cps. Unfortunately, present factory tests do not include a complete evaluation of tube response over a wide enough frequency spectrum.

Observations made on several equipments indicate that structural changes in the chassis or re-positioning of tubes would, in some instances, reduce the effect of mechanical disturbances on tubes. An occasional source of trouble is introduced by equipment motor vibrations, especially after mechanical wear has increased play in the moving parts. The accelerations involved in these vibrations are usually very small, but if their frequencies coincide with structural resonances in the tubes, unsatisfactory operation of the equipment may result. The influence of such disturbances on tube operation is not always recognized by equipment designers.

Several programs are currently pursued by both military and commercial agencies to increase tube reliability. Since the necessary requisites that make a tube reliable depend on the type of service to be performed and environmental conditions, the requirements stressed in the various programs differ in many respects. Some of these requirements are still in a state of flux, as actual needs are as yet not clearly known.

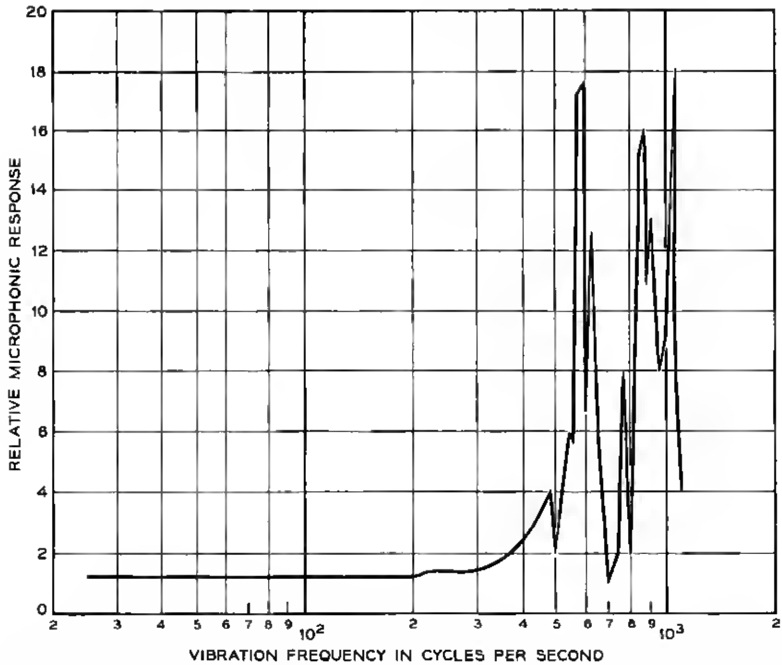


Fig. 9 — Frequency response of a tube that exhibited microphonism in equipment. The direction of vibration is perpendicular to the major diameter of grids and tube axis.

SHOCK AND VIBRATION REQUIREMENTS AND TEST EQUIPMENT

General requirements

Certain tests have been written into the MIL-E-1B[9] and other specifications to control the shock and vihration characteristics of tubes, and to check on their hehaviour under given mechanical excitation. Long usage has given these specifications some validity in that they control the quality of the product even though actual field conditions may not necessarily be simulated in the tests.

At present, one or more of the following three types of tests are called for on tuhe specifications:

(*a*) *Shock tests.* These are high acceleration tests to insure that tuhe structures will withstand occasional shocks of given maximum magnitudes. Since, in general, tubes are not required to function during high peak shocks, no operating voltages are applied to the tuhes for this test. The post shock requirements are that the tuhe characteristics must not have drifted out of their limits. In many cases, the type of shock tester to he used is specified, because it is difficult to define the shock output of

many of these machines in terms of acceleration and frequency content, etc. The response of tube structures may be vastly influenced by shocks of tbe same nominal acceleration but differing acceleration wave forms. Shock tests are also performed on tubes during their initial development to obtain the degree of cushioning required for safe sbipping and handling purposes.

(*b*) *Vibration tests*. These are low acceleration, fixed frequency tests. They are made on machines witb sinusoidal displacement output. Short time duration 25-cycle — 2.5g, or 50-cycle — 10g tests are specified for tubes that are generally not exposed to constant vibrations. Usually no voltages are applied to the tubes under test since their prime purpose is to check for sound tube structures. Electrical post vibration performances are tbe eriteria of tube quality. In tbis category also fall the long time duration vibration tests made on tubes intended for use in equipment that is known to be subjected to continual vibration, such as sbipboard or mobile equipment. Present specifications eall for 96-bour tests at 2.5g — 25 cycles. This, tberefore, is a fatigue test wbich determines the capability of tube structures, under prolonged cyclic stresses.

(*c*) *Microphonic vibration test*. A 25-cycle — 2.5 g test performed with specified voltages on the tubes under test to investigate the influence of low acceleration vibration on tbe output of tubes. This test is specified on certain tubes, especially those used for audio applications. The permissible magnitude of spurious signals excited by vibration is limited on tbe respective tube specifications.

(*d*) *Tap tests*. These tests are performed for two purposes. One is for the detection of defects such as foreign particles between close spaced adjacent elements, damaged elements, or poor welds. Open and short testers of given sensitivity are used to indicate these defects. The second purpose for tap testing is the investigation of microphonic response of tubes to mecbanical disturbances. For these tests tbe tubes are made to work in Class A amplifier circuits. Acoustic feedback may be used, so tbat the tube is not only subjected to the mecbanical tap, but also to the sound of the tap excited micropbonic signal which is reproduced tbrough a loud speaker spaced at a given distance from the tube. Sustained micropbonism can be produced by these means under certain eonditions. The purpose of this type of test is to simulate conditions to which tubes are subjected in some equipments, espeeially those closely coupled to audio output components.

Equipment

Tbe following is a brief description of the machines used for the performance of the above tests and their output characteristics.

MIL-E-1B Bump Tester

This is one of the earliest devices employed for shock testing of tubes under controlled conditions. In order to assure uniformity of results the MIL specifications give its physical dimensions. Fig. 10 illustrates the tester and its method of use. The magnitude of the shock and its duration is given by tube weight, shape of tube envelope contacted by the hammer, resilience of rubber pad on the hammer, and the angle (θ) through which the hammer is permitted to swing before striking the tube.

Although for the performance of the tests, only the angle (θ) is specified, the shock characteristics of this device have been investigated,[10] so that shock magnitudes and durations for any tube may be computed from the parameters given above. A typical acceleration time curve is shown in Fig. 11. The simple bell shaped outline of the accelerogram is given by the non-linear spring characteristic of the rubber pad and the generally cylindrical shape of the tube envelope.

Shock Testing Mechanism per ASA-C39.3

This mechanism is also used to check and compare the resistance of tubes to mechanical shocks of predetermined magnitude and duration. In this device the sample to be tested is rigidly fastened on a platform. A steel leaf spring supported at both ends is attached underneath this platform. The test on the sample is performed by raising the platform to a certain height, allowing it to fall on a steel anvil, and then catching it on the rebound.

The shock magnitude is given by

$$G \max = \sqrt{\frac{2hk}{W}}$$

and its duration by

$$J = \pi \sqrt{\frac{W}{12Kg}}$$

where W = tableweight (lbs.)
h = height of fall (inches)
K = spring constant of leaf spring.

A number of leaf springs are available to produce the desired shock characteristics. A full discussion of this mechanism and its performance are covered in Reference 11. A slightly modified version of the tester, used by the Laboratories, together with a typical shock pulse, is shown in Figs. 12 and 13. It can be seen that the pulse is essentially of sinusoidal shape with higher frequencies superimposed on it. The fundamental frequency is produced by flexing of the leaf spring during its contact

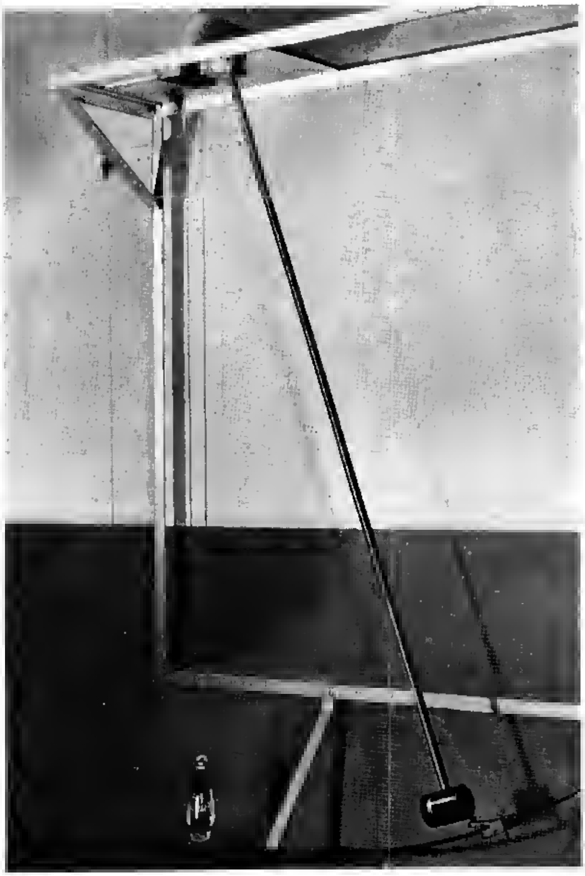


Fig. 10 — One of the early devices employed for shock testing of tubes under controlled conditions — the MIL-E-1B bump tester.

with the table anvil while the higher frequencies are due to table resonances which are excited by the metal-to-metal impact of spring and anvil.

High Impact Machine for Electronic Devices

This machine was originally intended to test the resistance of tubes to high peak-short duration shocks similar in nature to those encountered by equipment fastened to parts of ships that are likely to be exposed to direct explosion pressures.[12] The shock is produced by a steel hammer pendulum striking a movable steel table on which the tube under test is mounted. The shock magnitude is given by the angle through which the

pendulum is allowed to swing under the action of gravity before the hammer strikes the table. The forward motion of the table produced hy the impact is arrested by two shock absorbers.

The impact of the hammer on the anvil of the table produces an abrupt velocity change of the tahle and excites table resonances, both horizontally and vertically. Because of the structure of the table a very complex acceleration wave form results. In general the accelerations for a given hammer swing vary over the table surface. It is for this reason that, in testing procedures, the position of the tubes on the table and their method of clamping are well defined; and since the acceleration wave shape is the sum of many vibratory frequencies rather than a single shock pulse, the severity of the test is expressed by the angle of hammer swing instead of a shock magnitude and duration. Although some uniformity in performance is attained by rigid standardization of the structure of the machines and the above mentioned positioning of the tubes, minute differences in these parameters may produce sufficient variations in shock output to reflect on test results. Acceleration-time traces of shocks measured in the shock direction by an accelerometer fastened near the anvil of the table and a second accelerometer clamped to the center of the table, are shown in Figs. 14 and 15. The records were simultaneously taken through 10,000-cycle low pass filters. Even though the recording of accelerations containing high frequencies of large amplitudes is, to some extent, a function of the measuring equipment, it is seen that significant differences in output exist between the two points of measurements.

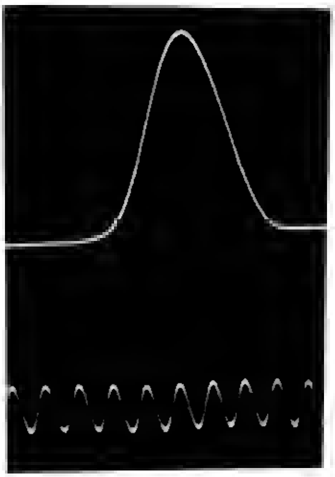


Fig. 11 — Acceleration-time pulse produced by MIL-E-1B bump tester.

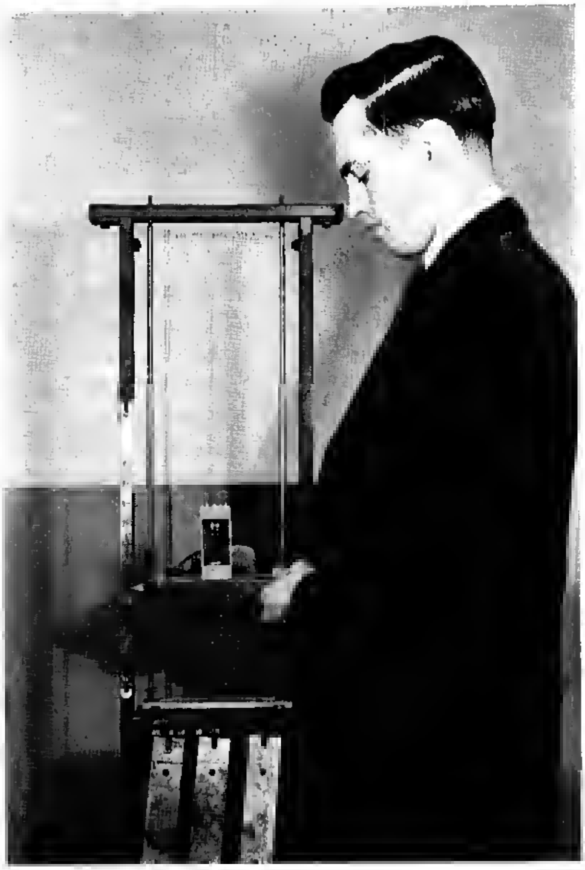


Fig. 12 — Shock testing mechanism per ASA-C39.3.

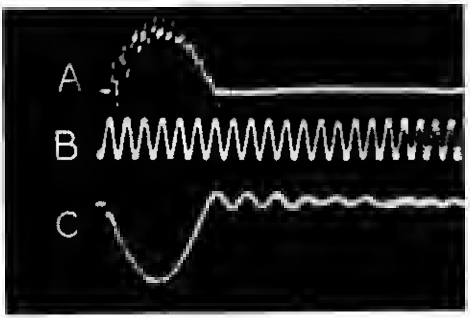


Fig. 13 — Typical shock pulse obtained with the tester shown in Fig. 12 (A) acceleration pulse, (B) timing trace, and (C) stress in leaf spring.

Fig. 14 — High-impact machine for testing electronic devices.

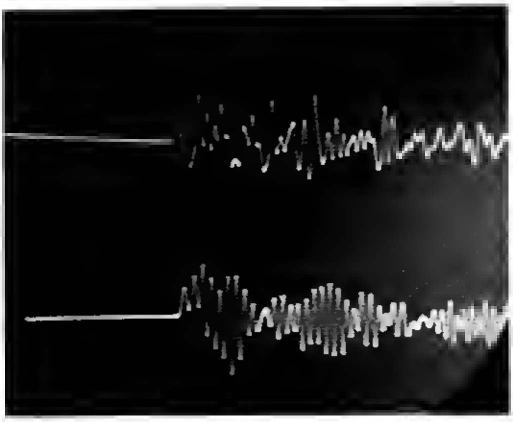


Fig. 15 — Acceleration-time pulses produced by machine shown in Fig. 14. Upper trace recorded near anvil, lower trace recorded in center of table.

This machine can also be used to produce lower G shocks of longer duration and simpler wave form by interposing a resilient rubber pad between the anvil and hammer. The resulting shock resembles the bell shaped output of the MIL-E-1B bump tester. Its magnitude and duration are given by the hammer swing, resilient characteristics of the pad, and the table and specimen weight. The capacity of the machine permits shock testing of heavier tubes (such as the larger magnetrons) by this method. Attainable shock levels are sufficiently high to cover the requirements placed on these tubes by conditions encountered during transit.

The L.A.B. Package Tester

Of particular interest to the packaging engineer, this machine is intended to duplicate the destructive vibrations and shocks experienced by a product during transit. It consists of a horizontal table that can be made to vibrate with a circular motion in the vertical plane. Adjustments permit variations of this motion to simulate freight car and motor truck movements. Tests performed on this machine, therefore, check on the mechanical strength of the outer shipping container as well as on the adequacy of cushioning materials employed to protect the product. The services are also considering this machine to test equipment designed for use in vehicles. Tests are now in progress by the Signal Corps to determine proper parameters for this application.[8]

Vibration Machines

A number of vibration machines, made by various manufacturers, are employed for vibration testing of tubes. Due to the high hash output of most mechanically driven machines, which contain gears and linkages, great care must be taken in the selection of these machines. Certain tests, especially the microphonic tests on receiving tubes, require machines with good sinusoidal output, in order to obtain comparable results. It is for this reason that the leaf spring vibration machine (Fig. 16), developed several years ago by Bell Telephone Laboratories, has been recommended as a standard for performing vibration tests. This is a fixed frequency, 25-cycle — 2.5 g machine which, due to its construction, has a relatively clean output as shown in Fig. 17.

Several types of electronically or motor-generator driven vibration machines are on the market. These are variable frequency and variahle amplitude machines especially useful for determining resonance frequencies of structures and for performing cycling vibration tests. Accessory equipment has been made available lately to conduct these tests on an automatic basis at either constant acceleration or constant am-

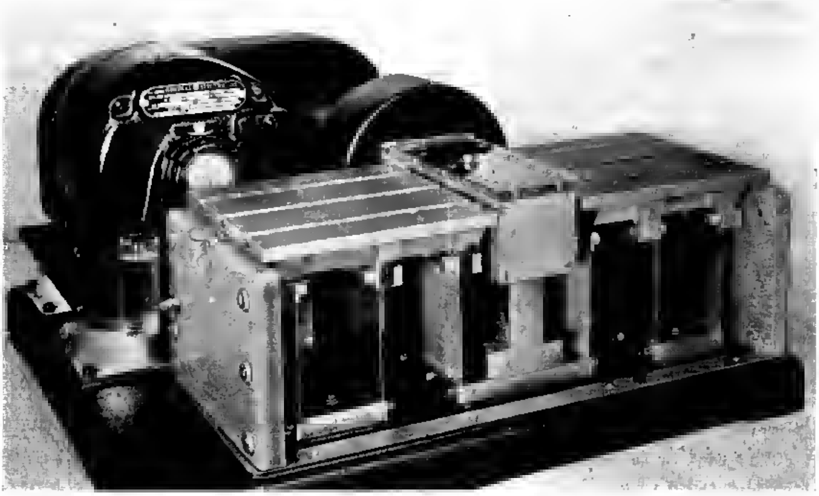


Fig. 16 — Leaf spring vibration machine developed several years ago by Bell Telephone Laboratories.

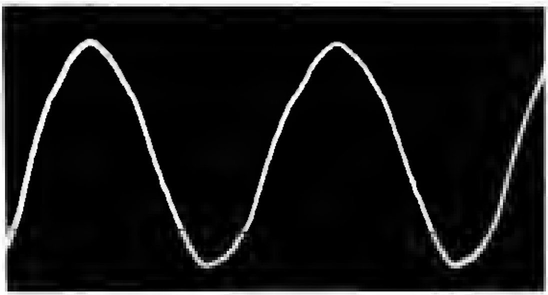


Fig. 17 — Acceleration output of the leaf spring vibration machine.

plitude over a given frequency range. Within the limitations of these machines and their power sources, it is also possible to subject specimens to complex disturbances. For instance, a projected use is the reproduction of recorded equipment vibrations so that the response of tubes to these conditions can be studied in the laboratory.

Tube Tappers

The physical construction of tube tappers used in the electron tube industry varies from the simple manually operated cork mallets to the more complicated automatic tappers. Since it has long been recognized that the reproducibility of manual tapping is rather poor, efforts have been made to replace these devices by automatic tappers or other

methods of checking microphonism in tubes. The industry, in collaboration with the Services, is now engaged in standardizing on automatic tappers. The prime requisites for such tappers are: (a) their shock output must be reproducible and must fall within definable limits, (b) their operation cycle must not adversely affect the time required for performing tap tests, and (c) the results obtained must have some relation to the environmental requirements for the tubes. This last condition is, perhaps, the most difficult to attain considering the diverse conditions encountered by tubes in various equipments.

Of the many tappers devised and employed by the industry, the use of only two are at present approved in the MIL-E-1B specifications. One is a manually operated cork mallet. This mallet is still retained in spite of its shortcoming, for lack of better devices. The second is the General Electric Automatic Tapper, specified for checking microphonism of some of the reliable tubes. This is a motor driven tapper which subjects the tube under test to damped low "g" vibrations at the rate of 2 taps per second. The tapper and associated circuits are fully described in the MIL specifications.

Space does not permit the discussion of other proposed tappers or test methods. Two rather interesting papers are listed in References 13 and 14, which illustrate the problems involved in the development of suitable devices for checking microphonism in tubes and describe some of the various methods of approach.

SUMMARY

The rapid growth of electronic equipment development during the late war has continued with the ever increasing new applications to both military and civilian purposes. It is realized that this growth has placed more stringent requirements on the mechanical characteristics of electron tubes. In order to define intelligently tube requirements, the nature of the disturbances to which tubes may be subjected must be known. It was pointed out that merely applying equipment requirements to the tubes, is not very realistic, since the shock and vibration patterns may be vastly modified by the equipment structures.

With the help of the recent development of light weight accelerometers, it is now possible to investigate the disturbances at the tube sockets. Both the Industry and the Services have begun to utilize these instruments to collect this information. The benefits derived from this work will enable the equipment and tube designers to formulate more accurate requirements and to devise test gear that will simulate more closely field conditions to check on tube quality.

Although, as a result of the trend to miniaturization, the strength of electron tubes has been increased, due to the smaller size and mass of elements employed, much has still to be done to increase tube resistance to ballistic shock. This would result in simplification of shipping containers and reduce the need for protection by shock absorbers in equipment. Equally important is the effort now made to reduce the microphonic response of the tubes. Here, too, the smaller size of the late tubes is of advantage because element resonant frequencies are increased. And since the higher frequency components of disturbances are largely attenuated by equipment structures, the tube responses have been lessened.

With a better understanding of the problems involved in tube protection, it is also quite possible that further improvements can be effected in some instances by structural changes of equipment members, or re-orientation of tubes in the equipment to reduce the effects of shock and vibrations on tubes.

BIBLIOGRAPHY

1. Shock and Vibration Instrumentation, NRL report #0-2645.
2. Mechanical vibrations, Den Hartog, McGraw-Hill Book Co.
3. The reciprocity calibration of piezo-electric accelerometers, M. Harrison, A. O. Sykes, and P. G. Marcotte, The David Taylor Model Basin, Report 811.
4. A Noise Free Instrument Cable, National Bureau of Standards, Technical News Bulletin, **36**, No. 3.
5. Research and Development Toward Construction of Accelerometers, Gulton Mfg. Co. report. New Techniques for Measuring Forces and Wear, W. P. Mason, B.S.T.J., **31**, May, 1952. Proceedings of symposium on Barium Titanate Accelerometers, National Bureau of Standards, Report No. 2654, Aug. 1953.
6. Bureau of Ships Publications, Nav. Ships 900,036, and Nav. Ships 900,052. The David W. Taylor Model Basin, Report 481. Vibration and Shock Isolation, C. E. Crede, John Wiley & Sons. Response of Damped Elastic Systems to Transient Disturbances, R. D. Mindlin, F. W. Stubner, H. L. Cooper, Experimental Stress Analysis, **5**, No. 2. Measured Aircraft Vibrations as a Guide to Laboratory Testing, D. C. Kennard, Jr., A. F. Technical Report No. 6429.
7. Dynamics of Package Cushioning, R. D Mindlin, B.S.T.J., **24**, July — Oct., 1945.
8. Evaluation of Electron Tube Quality on the Tethered Weapons Carrier, Signal Corps Engineering Laboratories, Army Project No. 0302A.
9. MIL-E-1B, Military Specifications for Electron Tubes.
10. Characteristics of the JAN Bump Tester, R. D. Mindlin, F. W. Stubner and H. A. Lefkowitz, MM-45-2920-4.
11. Shock Testing Mechanism for Electrical Indicating Instruments, ASA-C 39.3
12. Mechanical Shock Characteristics of the High Impact Machines for Electronic Devices, I. Vigness, R. C. Novak and E. W. Kammer, NRL Report No. 0-2838.
13. Proposal for Revised Noise and Microphonic Evaluation for MIL-E-1B Specifications, NML Report, Dated July 30, 1951.
14. A Study of Microphonism, Philco Research Report No. 183.